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EXECUTIVE SUMMARY

MODELLING UNBOUNDED CRITICAL TWO-PHASE FLOW CRITICAL DISCHARGE EXPERIMENTS

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This report describes a study of unbounded critical two-phase flow which may occur if a reactor header or feeder ruptures. The pressurized liquid will escape from the break and flash into a mixture of steam and drops. Fission products in the coolant will be carried into containment by this jet. It is important to know the characteristics of this jet to assess the transport of fission products through containment. As a prelude to studying and modelling the jet, this report describes the evaluation of critical discharge models and photographs of the jet.

Measurement of critical discharge was performed using the Thermo-Fluids pressurized water blowdown facility. This facility is described in report 87-320-K. The experimental method is described in this report. Three correlations are compared with the experimental data. These include the Homogeneous Equilibrium model, the Henry-Fauske model and the Alamgir-Lienhard correlation. The Henry-Fauske model gave the best agreement with the data and the model is relatively easy to apply.

The jet was photographed for a variety of reservoir conditions. Three regions were identified: (1) a high density cone or dart, (2) a lower density region with the same expansion angle, and (3) a region with a smaller expansion angle.

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To Dr. D. Mills
Director
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MODELLING UNBOUNDED CRITICAL TWO-PHASE FLOW
CRITICAL DISCHARGE EXPERIMENTS

1.0 INTRODUCTION

The Thermo-Fluids Unit of the Mechanical Research Department is studying unbounded critical two-phase flows. This phenomenon is found in a LOCA when pressurized coolant escapes from a ruptured feeder or header. The escaping liquid flashes and a mixture of steam and droplets is propelled into the containment system. Fission products in the coolant are transported into containment by this two-phase jet. It is important to know how the fission products are transported through containment so that releases from containment to the environment can be properly assessed. There are many pathways for transport and depletion of fission products in containment, but all pathways are linked to the coolant system by the critical discharge and the flashing process. There has been much study to date on critical discharge from nozzles, orifices and short tubes. The purpose of this study is to evaluate existing critical discharge models and to analyse the flashing process with the aid of photographic data.

2.0 MEASUREMENT OF CRITICAL DISCHARGE

Critical discharge experiments were performed using the Thermo-Fluids pressurized water blowdown facility. This facility is described in report 87-320-K. Some highlights of the facility are its ability to produce flashing at the throat of a short converging nozzle and to measure the thermodynamic conditions immediately upstream of the nozzle. The entire blowdown facility is mounted on a weigh scale and

total discharge can be measured to ±0.05 kg. A wide range of subcooled conditions up to 3 MPa can be maintained in the pressure vessel.

Following is the operating procedure:

1. All hydraulic ports were closed, the pressure vessel was pressurized to ≈5 kPa with nitrogen, and the drainage port was opened. The source of nitrogen was shut off when it was observed that no more water was leaving the drainage port. The drainage port was closed. After a few minutes, the weigh scale was reset to zero. The vent line was opened and deionized water was loaded into the pressurized vessel. Filling continued until water flowed from the vent line. Then the valve at the filling port was closed. This step took ≈1/2 hour.

2. The heater control was set to 100°C. The heaters were turned on by setting the heater override dial to a temperature somewhat above the set point and depressing the reset button. The vent line was closed a few minutes after boiling had commenced. This step took ≈1/2 hour.

3. The heater override dial was set to a value somewhat above the desired blowdown temperature. Then the heater control was set to the blowdown temperature. It took ≈3/4 hour to raise the pressure vessel temperature from 100°C to 230°C.

4. While the water was heating up, the data acquisition computer was set up to monitor and display the pressures and temperatures. A grid was mounted over the nozzle, the camera was mounted on the stand and the grid was photographed.

5. When the water was at the set point temperature, the heaters were shut off by rotating the the override dial to 0°C. Then the temperatures, pressures and mass were recorded.

6. The data acquisition computer was set up for logging and started. Then the valve upstream of the nozzle was opened. During this time, photographs were taken of the jet using either flash or back lighting. The camera was not moved from the stand during the blowdown.

7. The blowdown was allowed to proceed until ≈15 kg had discharged from the pressure vessel. Then the valve upstream of the nozzle was closed and the data acquisition computer was instructed to stop recording and store the data onto floppy disk. Finally the mass as displayed on the weigh scale was recorded.

The discharge time, reservoir pressure and reservoir temperature were obtained from the computer-logged data. Figure 1 shows typical pressure and temperature histories for one blowdown. It can be seen that the pressures decayed somewhat during the blowdown. The nozzle entrance pressure was higher than the vessel pressure. This was determined to be a calibration error. Experience showed that the nozzle pressure transducer was more reliable. The discharge time was determined from the nozzle pressure history where it can be seen the pressure rises and falls rapidly at the opening and closing of the discharge valve. The discharge pressure was taken as the average of all logged pressures over the discharge time. Three reservoir temperatures were recorded: (1) in the pressure vessel at the elevation of the vent line; (2) close to the pipe entrance to the valve assembly and; (3) immediately upstream of the nozzle. These temperatures were labelled: (1) vapour temperature; (2) pipe entrance temperature and; (3) nozzle inlet temperature, respectively. The vapour temperature was initially higher than the pipe inlet temperature but this difference diminished as the blowdown proceeded. This difference was attributed to temperature stratification in the pressure vessel. The nozzle inlet temperature was significantly lower than the pipe inlet temperature. This difference can be attributed to transient heat loss to the valve assembly and steady-state heat loss to the laboratory air through the valve assembly. To reduce the heat

loss, the valve assembly was insulated by wrapping it with asbestos cloth. Figure 2, shows that the insulation reduced the difference between the pipe inlet and nozzle upstream temperatures.

When a large subcooling was required, the pressure vessel was pressurized with nitrogen. The following procedure was used in concert with the above procedure:

4a. The mass as shown on the weigh scale was recorded. While monitoring the vessel pressure via the computer monitor, the nitrogen regulator was adjusted to give the desired pressure.

6a. The nitrogen pressure regulator was periodically adjusted to maintain the vessel pressure at the desired set point.

7a. After the water had cooled below 100°C, the nitrogen cover gas was vented and the mass was recorded.

In the event that the mass discharge without N_2 was much different than that with N_2, the latter was chosen. The reason for the difference may be that steam or water escaped from the discharge valve during cool-down.

A second series of tests was performed to determine the discharge coefficient for the nozzle when the liquid did not flash. In this case the DI water was not heated up but the nitrogen cover was used for pressure control. A similar operating procedure was followed with the exception that no photographs were taken. The results of these tests are shown in Table 1.

Table 1
DISCHARGE COEFFICIENT - COLD FLOW

FILE	PRESSURE [MPa]	MASS FLUX [kg/m^2/s] ideal	measured	C_D	Re
UCRIT11	3.0	75987	66400	0.87	66400
UCRIT16	1.0	42346	36200	0.85	36200
UCRIT18	0.4	24267	21400	0.88	21400
UCRIT20	0.5	28202	24700	0.88	24700
UCRIT21	0.7	34565	30800	0.89	30800
UCRIT22	0.9	39939	34600	0.87	34600
UCRIT23	1.1	44662	39900	0.89	39900

3.0 COMPARISON WITH EXISTING CORRELATIONS

The literature was reviewed to find critical discharge models for initially subcooled water. This review uncovered three models which were compared with the experimental data from this study. These models are first described. Following that is a commentary on the models.

3.1 HOMOGENEOUS EQUILIBRIUM MODEL

This is the simplest analytical approach and it is described in detail by Starkman et al [1]. This model is based upon the following assumptions:

1. The velocities of the two phases are equal.
2. Thermal equilibrium exists.
3. The expansion is isentropic.
4. Properties data correspond to those of a static, equilibrium two-phase system with plane interfaces.

The flow prediction resulting from these assumptions is:

$$G_{cHE} = \frac{\{2[h_o - (1 - x_E)h_{\ell E} - x_E h_{gE}]\}^{1/2}}{(1 - x_E)v_{\ell E} + x_E v_{gE}}$$

The critical flow rate prediction is computed by choosing successively lower downstream pressures until the flow rate exhibits a maximum.

3.2 HENRY-FAUSKE MODEL

This model assumes there is a deviation from thermodynamic equilibrium, i.e. the liquid superheats and the vapour supersaturates [2]. This assumption is expanded in the following points:

1. Equal velocities for the vapour and liquid phases.
2. No heat or mass transfer during the expansion to the throat.
3. Isentropic expansion of each phase.
4. Rates of heat and mass transfer at the throat are significant and can be represented by simple correlations.

The expression for the mass flux is derived from the two phase momentum equation. At critical flow, the mass flow rate exhibits a maximum with respect to pressure. This derivative is applied to the momentum equation. The resulting expression contains first derivatives of specific volume, quality and slip with respect to pressure. For the case of initially saturated or subcooled liquid, only the term with the derivative of quality with pressure remains. It is assumed that this derivative can take the following form:

$$\left.\frac{dx}{dP}\right|_t = N \left.\frac{dx_E}{dP}\right|_t$$

where $N = N(x_E)$

From experimental results, the following result was determied:

$$N = x_{Et}/0.14$$

For throat equilibrium qualities greater than 0.14, N is set equal to unity.

Combining these expressions with the mass flux equation yields:

$$G^2 = \left[(v_{gE} - v_{\ell o}) \frac{N}{(s_{gE} - s_{\ell E})} \frac{ds_{\ell E}}{dP} \right]^{-1}$$

Since no vapour is formed until the throat, it is assumed that the critical pressure ratio can be calculated from the single phase momentum equation. The critical pressure ratio can thus be calculated as follows:

$$\eta = 1 - (v_{\ell o} G^2/2P_o)$$

The above two equations can be combined to give a trancendental expression for the critical pressure ratio. Having solved for this, the mass flux can be calculated.

3.3 PRESSURE UNDERSHOOT CORRELATION

The pressure undershoot correlation was developed by Alamgir and Lienhard to predict static decompression undershoot of initially saturated or subcooled liquid [3]. However, as suggested by Abuaf et al, the decompression rate Σ', utilized in the correlation, can be set in the Eulerian frame [4]. Consequently, the correlation can be applied to flow in a converging nozzle. The correlation is as follows:

$$P_{sat} - P_n = 0.252 \frac{\sigma^{3/2} T_r^{13.73} (1 + 14\Sigma'^{0.8})^{0.5}}{\sqrt{kT_c}} \left[1 - \frac{v_\ell}{v_g} \right]$$

The depressurization rate can be set in the Eulerian frame as follows:

$$\Sigma' = \left.\frac{\partial P}{\partial t}\right|_z + V \left.\frac{\partial P}{\partial z}\right|_t$$

In this application, the reservoir conditions are constant so the mass flux is constant. Consequently, Σ' is dependent only on the convective term in the above equation and not on the temporal term. The convective depressurization was calculated from the momentum and continuity equations for incompressible, single phase flow. The resulting expression is:

$$\frac{\partial P}{\partial z} = \rho \frac{(V_t A_t)^2}{[A(z)]^3} \frac{dA}{dz}$$

This expression was multiplied by the local velocity as calculated from continuity to give:

$$\Sigma' = V \frac{\partial P}{\partial z} = \rho \frac{(V_t A_t)^3}{[A(z)]^4} \frac{dA}{dz}$$

where V = velocity
A = area
t = conditions at the throat

To calculate the critical discharge, the maximum depressurization rate was required. From the above expression, it can be seen that this required the maximization of the area terms. Fortunately, the

cross-sectional area of the nozzle was known (see Figure 3). The area terms were maximized with respect to z, to give:

$$\left.\frac{dA/dz}{[A(z)]^4}\right|_{max} = 5.582 \cdot 10^{20}\ m^{-7} \quad @\ z = 0.6\ mm$$

Given these expressions, it was possible to calculate the mass flux from the following iterative procedure:

1. Estimate the throat pressure.
2. Calculate the throat velocity from the single phase momentum equation and the above estimate.
3. Calculate the depressurization rate from the derived expression and the throat velocity.
4. Calculate the pressure undershoot using the Alamgir-Lienhard correlation.
5. Use the pressure undershoot as the throat pressure in the momentum equation to calculate the throat velocity. Compare this velocity with that calculated in step 2.
6. Repeat steps 1 to 5, choosing sucessively lower throat pressures until convergence is acheived on the throat velocity.

The mass flux was calculated from the throat velocity and the density of saturated water based on the reservoir temperature.

3.4 EVALUATION OF CRITICAL DISCHARGE MODELS

The three critical discharge models described above were compared with the experimental data (see Table 2). Figure 4 shows calculated and experimental data for reservoir pressures 1.0 & 3.0 MPa. The pressure undershoot (Alamgir-Lienhard) model overpredicts the data; while the HEM model underpredicts the data. Overall, the Henry-Fauske model provides the best agreement with the data. This model appears to work best for large subcoolings.

Table 2
MASS FLUX THROUGH NOZZLE

File	Press. [MPa]	Temp. [°C]	$(T_{sat}-T_o)/T_{sat}$	MASS FLUX [kg/m²/s]			
				Experiment	Henry	Alamgir	HEM
UCRIT03	1.0	173	0.0384	27000	20721	26072	14384
UCRIT15	1.0	160	0.1107	30900	27613	32445	16417
UCRIT17	1.0	152	0.1552	31700	30856	35019	28638
UCRIT19	1.0	143	0.2052	32500	33846	37219	28512
UCRIT10	1.1	179	0.0277	27800	21491	26655	11806
UCRIT04	1.6	193	0.0418	33000	28030	35940	16698
UCRIT05	2.0	202	0.0490	36000	32175	42759	23096
UCRIT06	2.6	219	0.0313	41000	33083	49403	21990
UCRIT09	3.0	229	0.0214	44100	34064	53077	19780
UCRIT07	3.0	227	0.0299	43000	35318	54704	22675
UCRIT12	3.0	207	0.1150	51600	48115	63580	37712
UCRIT13	3.0	196	0.1620	52800	53598	66456	44743
UCRIT14	3.0	185	0.2090	53700	58682	68487	47832

Abuaf et al described a modification to the pressure undershoot model which provides better prediction of critical discharge [4]. The proposed method includes a discharge coefficient in the calculation of the mass flux. Since it is assumed that the flow is single phase up to the throat, the discharge coefficient should be the same as that for cold discharge. Figure 5 shows that for a similar range of mass flux, a C_D of 0.85 will give good agreement with experimental data, in particular, for mass flux greater than 30,000.

4.0 PHOTOGRAPHS OF UNBOUNDED CRITICAL TWO-PHASE FLOW

The flashing jet was photographed to identify regions in the jet and to measure the major geometric parameters. Illumination by flash was useful to see the outline of the jet; while illumination from behind the jet revealed structures within the jet. Backlighting was

accomplished with an overhead projector. A problem with this method of illumination was that the light was not uniform in intensity. Figures 6 & 7 show a jet of initially subcooled water at 2.6 MPa and 219°C. The jet appears to expand initially at 45° and then taper down to a much smaller angle. Three regions can be identified: (1) a high density cone; (2) a region of lower density where the expansion angle is the same; and (3) a region with a smaller expansion angle. Using smoke as a tracer, it was found that air entrainment was important in the latter two regions. Figures 8 & 9 show a jet of initially subcooled water at 3.0 MPa and 229°C. The above mentioned geometries are similar but the initial expansion angle is closer to 55°. The next two figures (10 & 11) show a jet of initially subcooled water at 1.1 MPa & 179°C. The initial expansion angle was close to 35° and gradually tapered to 20°. The high density cone in region 1 was replaced by a high density cylinder.

5.0 CONCLUSION

Critical discharge models have been evaluated for the short converging nozzle described herein. The Henry-Fauske model provides the best agreement with the experimental data. Given the discharge coefficient for the nozzle, the Alamgir-Lienhard correlation may also be applied to predict critical discharge. If the nozzle discharge coefficient is included in the calculation, this procedure also provides accurate predictions. The disadvantages are that (1) it is specific to the nozzle geometry and (2) the calculation is more complex than the Henry-Fauske model. Both models have been applied to subcooled discharge only.

Photographs of the jet were useful to identify its general structure. This information can be used for deciding the locations for detailed study. The backlighting can be improved by using a light source of uniform intensity.

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List of Symbols

C_D = discharge coefficient
Re = Reynolds number
G = mass flux
h = enthalpy
P = pressure
P_n = presure undershoot according to Alamgir-Lienhard correlation
N = Henry-Fauske non-equilibrium coefficient
s = entropy
T = temperature
T_r = reduced temperature (T/T_c)
T_c = critical temperature
t = time
k = Boltzman's constant
z = axial coordinate
A = cross-sectional area
V = velocity

Greek

x = quality
η = critical pressure ratio
Σ' = Lagrangian decompression rate
σ = surface tension
ρ = density

Script

$\mathcal{v}$ = specific volume

Subscript

c = critical conditions ie pertaining to choked flow

HE = pertaining to Homogenous Equilibrium model

o = stagnation or reservoir conditions

E = thermodynamic equilibrium

g = gas phase

ℓ = liquid phase

t = at the nozzle throat

sat = thermodynamic saturation conditions

max = maximized with respect to the z or axial coordinate

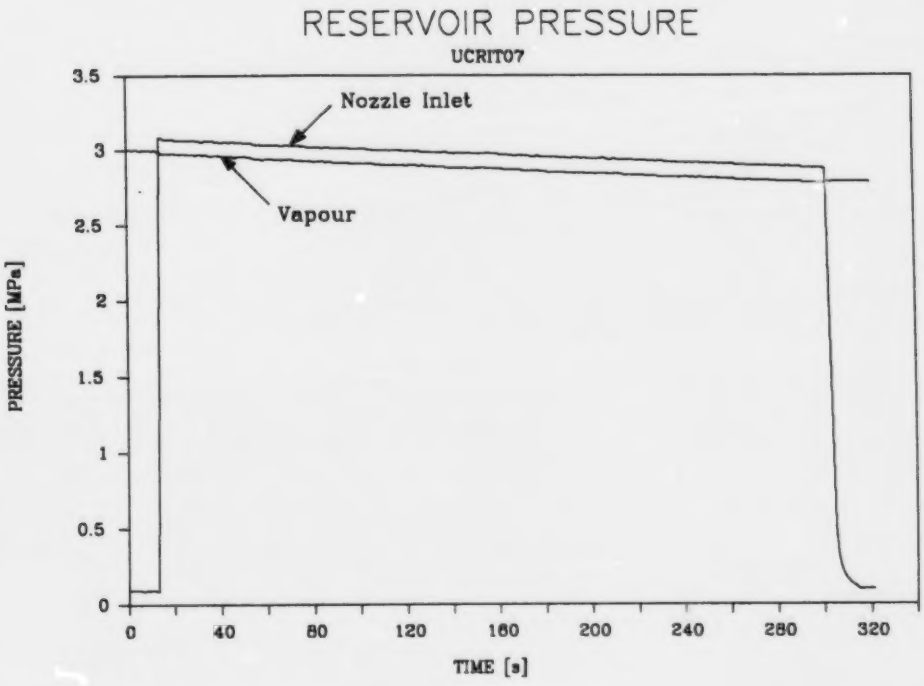
References

1. E.S. Starkman, et al, "Expansion of a Very Low Quality Two-Phase Fluid Through a Convergent-Divergent Nozzle", Journal of Basic Engineering, June, 1984.

2. R.E. Henry, H.K. Fauske, "The Two-Phase Critical Flow of One Component Mixtures in Nozzles, Orifices, and Short Tubes", Journal of Heat Transfer, May, 1971.

3. M. Alamgir, J.H. Lienhard, "Correlation of Pressure Undershoot During Hot-Water Depressurization", Journal of Heat Transfer, Vol. 103, February, 1981.

4. N. Abuaf, et al, "Critical Flashing Flows in Nozzles with Subcooled Inlet Conditions", Journal of Heat Transfer, Vol. 105, May, 1983.



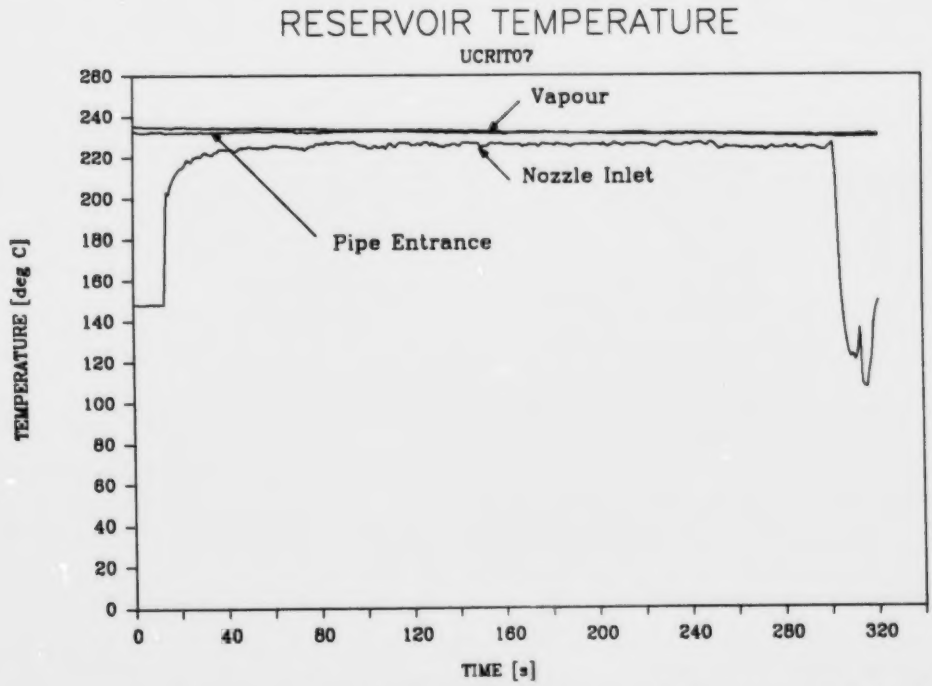
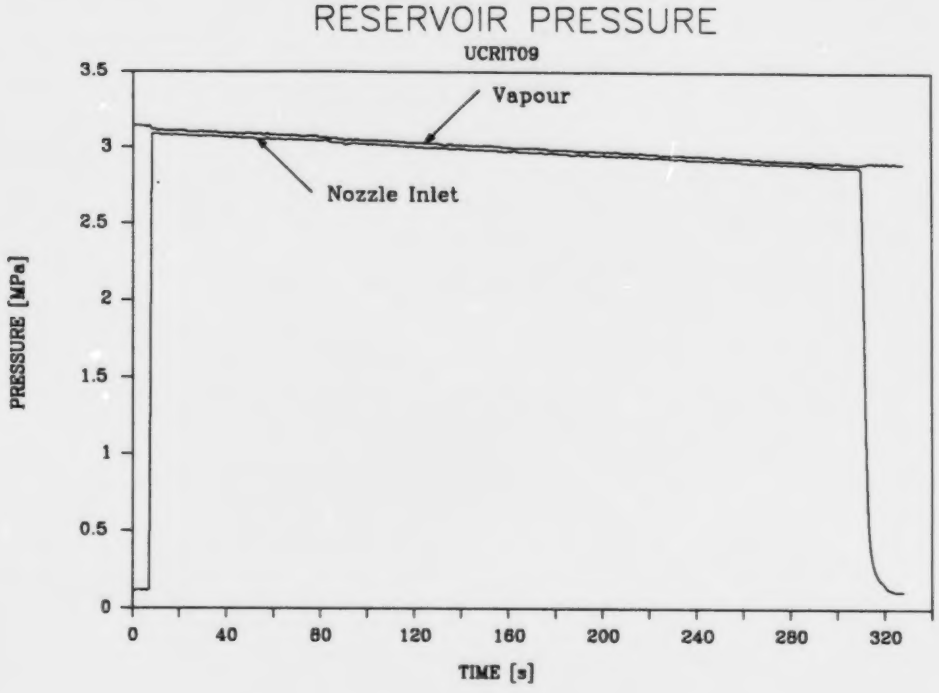


Figure 1 – Typical Temperature and Pressure Histories



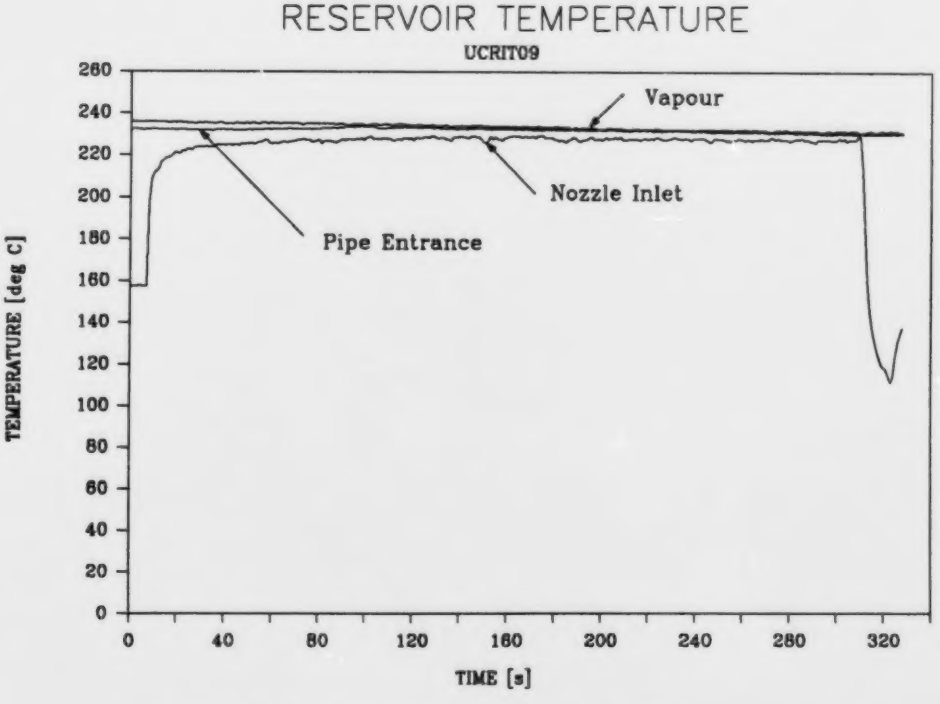


Figure 2 – Use of Insulation to Reduce Heat Loss

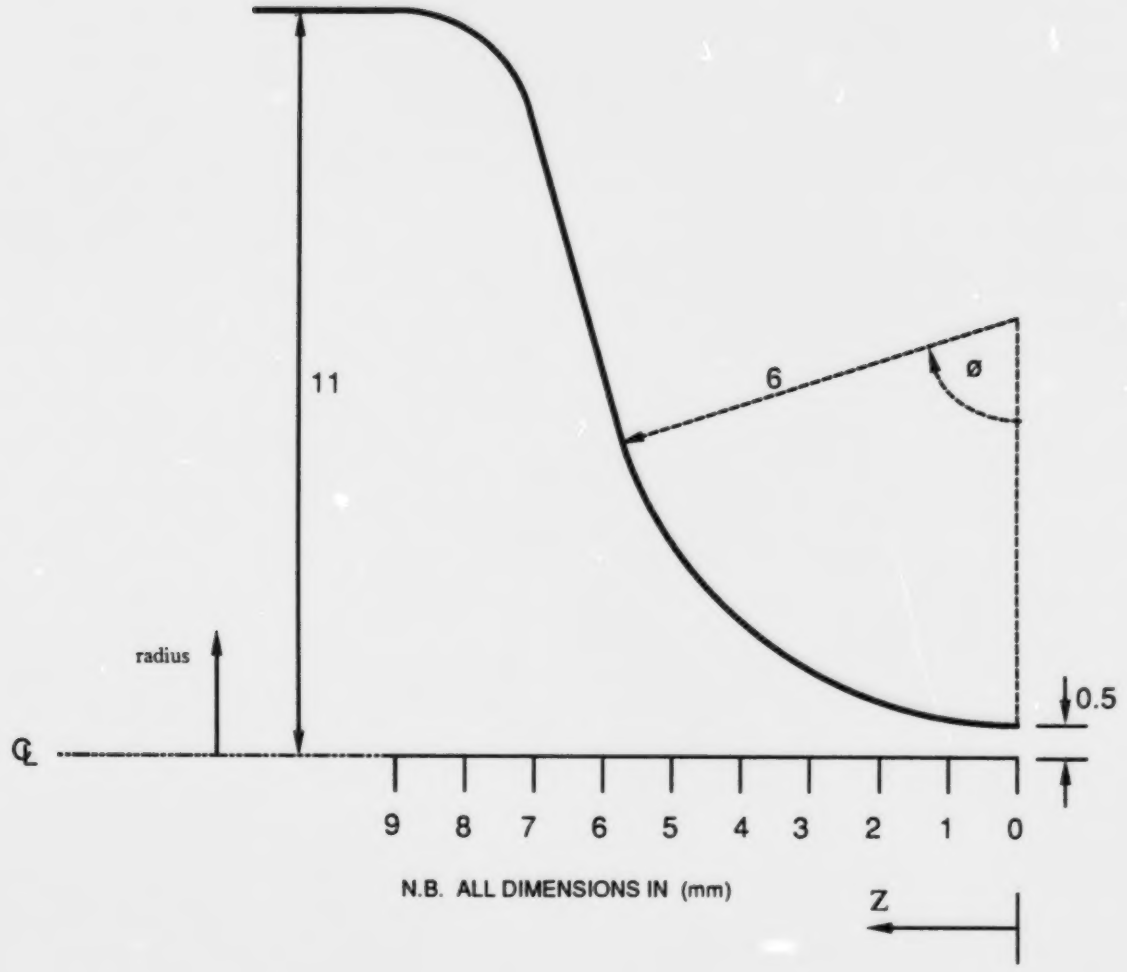


Figure 3 : Cross section of Nozzle

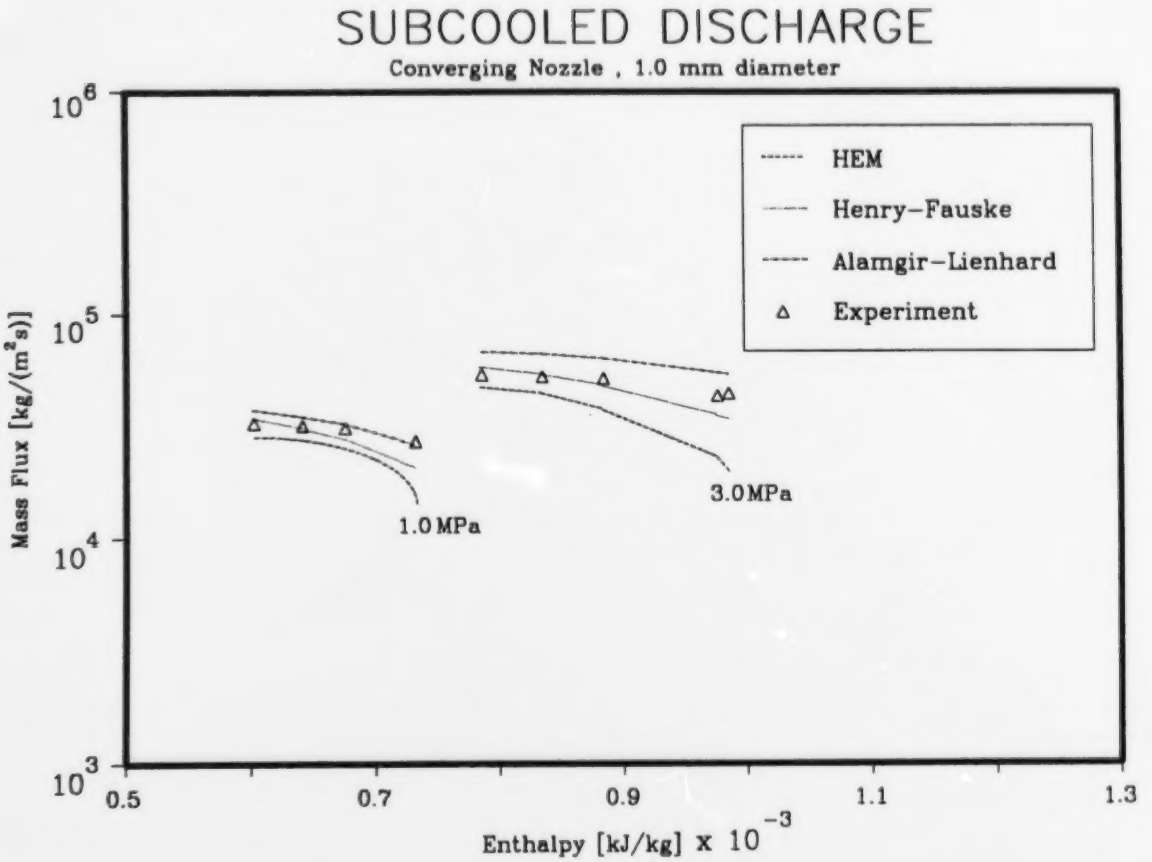
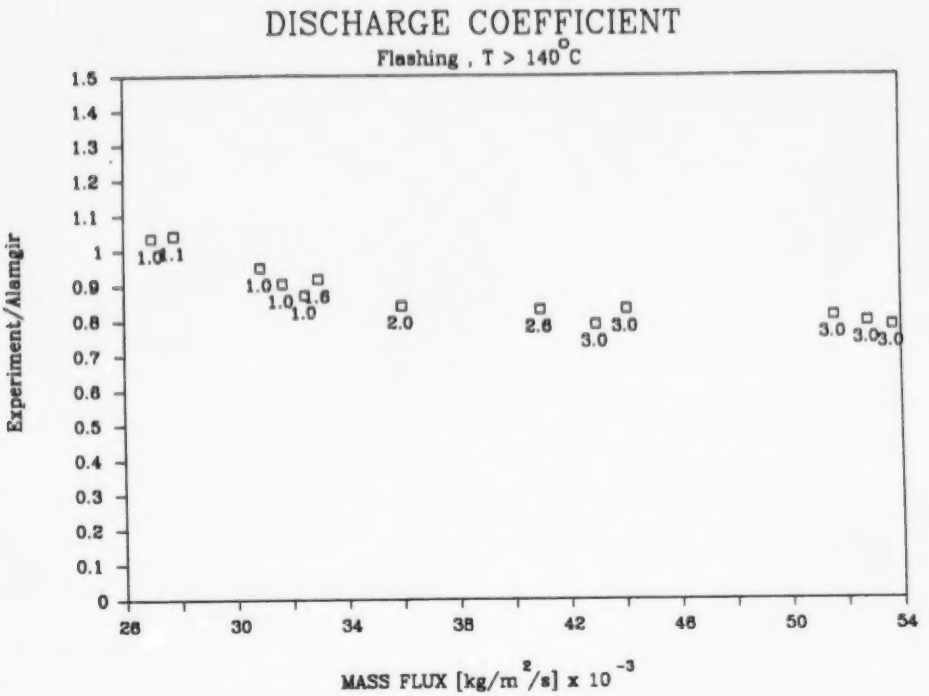


Figure 4 : Critical Discharge of Initially Subcooled Water



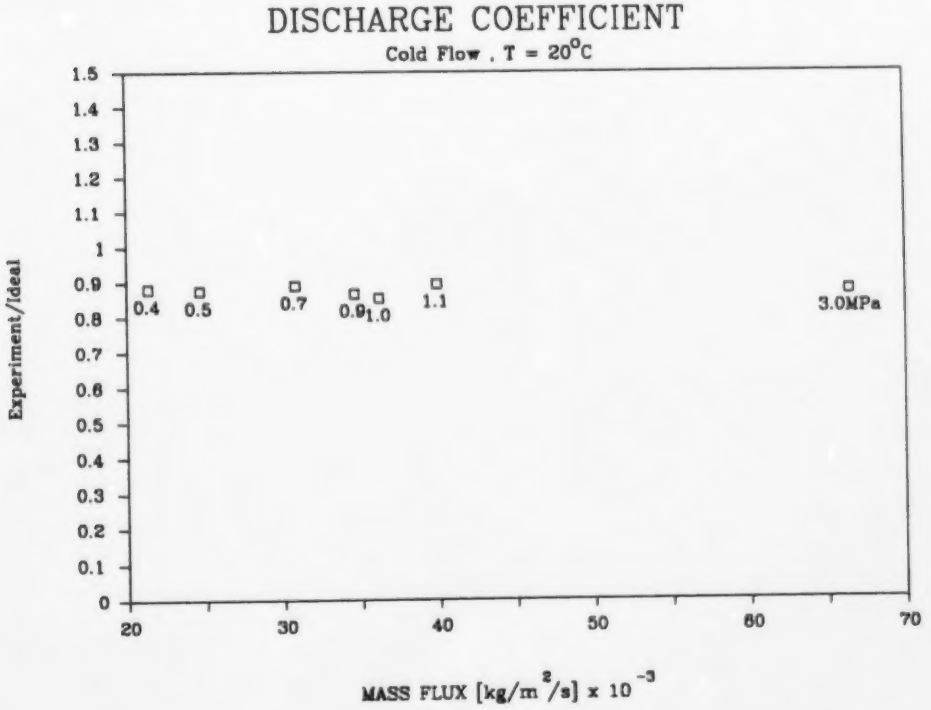


Figure 5 : Discharge Coefficient for Subcooled Water

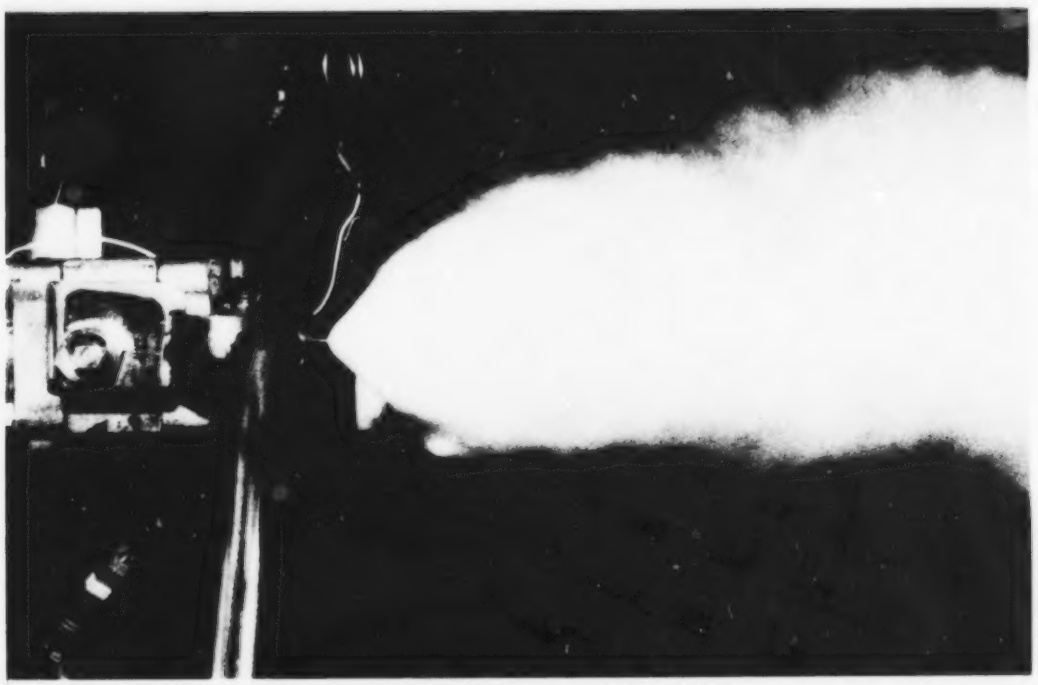


Figure 6 : Two-Phase Jet , 2.6 MPa , 219°C (flash illumination)

Figure 7 : Two-Phase Jet , 2.6 MPa , 219°C (backlighting)

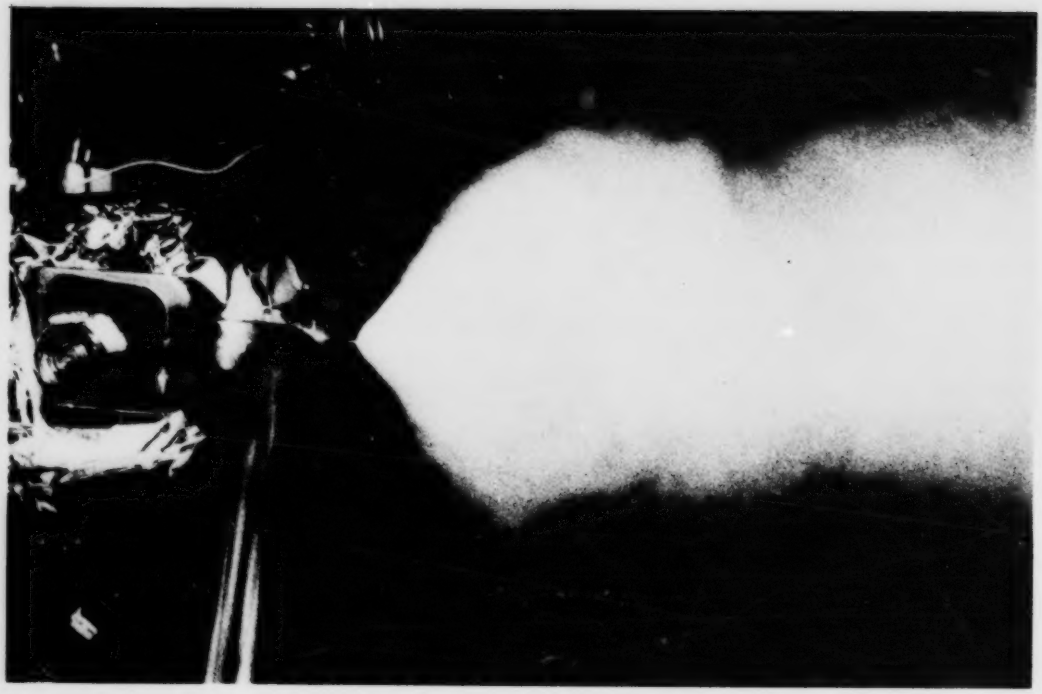


Figure 8 : Two-Phase Jet , 3.0 MPa , 229°C (flash illumination)

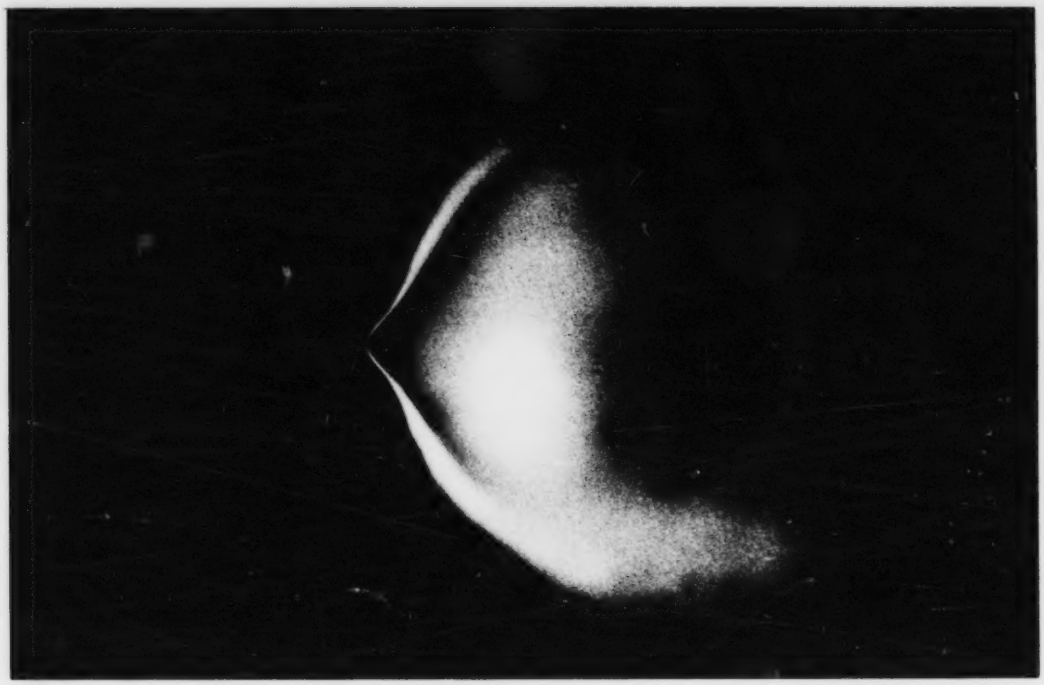


Figure 9 : Two-Phase Jet , 3.0 MPa , 229°C (backlighting)

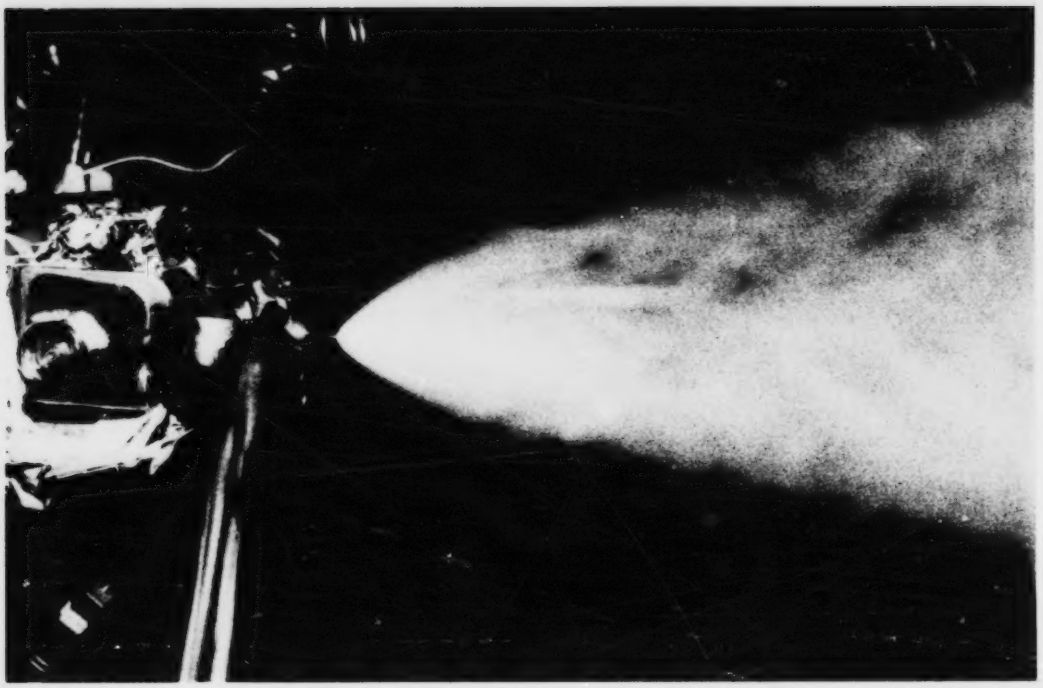


Figure 10 : Two-Phase Jet , 1.1 MPa , 179°C (flash illumination)

Figure 11 : Two-Phase Jet , 1.1 MPa , 179°C (backlighting)